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Smoke Filling in an Enclosure*

*The filling of an enclosure by smoke generated by a diffusion flame was studied by
measuring the smoke extinction coefficient and temperature versus height during
the filling process. Semi-quantitative agreement is found between the Baines-Turner
theory and experiments with the burner located in the center of the room for heat
fluxes over the range 11 to 32 kW. The filling process was found to be about twice
as fast when the burner was moved off center. A novel technique for generating a
thin smoke layer is described. Approximately 80 percent of the total heat generated
by combustion was lost to the ceiling and walls. It was found that the conversion
ratio of fuel to particulate is a non-linear function of fuel flow rate and that the
shape of the curve is sigmoidal, which appears to be a common characteristic of
sooting systems.*

INTRODUCTION

Smoke movement is a subject of great interest in
fire safety because of the hazardous effects of smoke
and because of its ability to provide early warning
of fire threat. Here the term smoke is used to
include both the particulate and gaseous components
of the combustion products. Smoke affects visibility
thus making evacuation and rescue from a developing
fire environment more difficult. In a majority of
fire related deaths in buildings, the cause of death
is attributed to smoke inhalation with the CO compo-
nent of the smoke often the immediate cause of death.
The fact that smoke can readily be detected is the
basis of most fire protection systems. For example,
smoke detectors are sensitive to the particulate com-
ponent of smoke and sprinkler systems are activated
by the high temperature of the smoke gases.

The movement of the smoke from the point of
origin to other locations will determine the time at
which the presence of the smoke is detected and the
time at which the conditions become hazardous. The
general subject of smoke movement in a building can
be broken down into two parts. One part concerns the
movement of smoke in a single enclosure and the other
is concerned with the movement of smoke from the room
of fire origin to other rooms in the building. This
study is concerned with the movement of smoke in a
single room size enclosure with no forced ventilation.

Baines and Turner[1] developed a theory of turbulent
buoyant convection from a source in a confined region.
This theory, together with variants of it developed by
a number of researchers including Kawagoe[2] and
Zukoski[3], has been applied to the hot smoky region as
it moves from the ceiling toward the floor. However,

there has been no detailed study of the range of
validity of this theory for combustion generated
plumes. The verification of the theory rests on salt
water plume experiments. Because of the possible
effect of heat transfer to the walls and ceiling in
the case of the gaseous plume, there is a need for
smoke and temperature measurements during the "fil-
ling" of the enclosure with smoke to determine the
range of validity of the theory. The study presented
here consists of such measurements for plumes gener-
ated by a diffusion burner for a room size enclosure.
The gaseous plume experiments are more difficult than
the salt water experiments because of the larger
scale, faster filling time (on the order of ten times
faster), and inherent difficulty in measuring the
position of a smoke/air interface compared to a thin
dye layer in the salt water experiment.

In the next section the key results of the theory
developed by Baines and Turner and by Zukoski for a
plume in an enclosure are expressed in a form appli-
cable to a gaseous plume containing smoke particles.
Section 3 contains a description of the instrumenta-
tion used for measuring the temperature and smoke
concentration during the movement of the smoke/air
interface. The key features of the diffusion gas
burner used for this study are steady state output,
repeatability in terms of heat flux and particulate
mass flux, and flexibility in terms of fuel flow rate
(11 to 32 kW) and type of fuel (propane and acety-
lene).

The principal results of this study presented in
Section 4 include the position of the smoke/air inter-
face versus time, the effect of burner position on
the smoke "filling" process, the smoke concentration
and temperature profiles, and the variation of the
average smoke concentration and temperature with
respect to time. These results are compared with the
theory developed by Baines and Turner and by Zukoski
and with the experimental results of Miyamoto et al.[4]
and Kawagoe[5] in Section 5. Both of these experimental
studies involved smoke "filling" in room size enclo-
sures using methanol pool fires with smoke flares to
allow visual observation of the smoke/air interface.

SUMMARY OF THEORETICAL RESULTS FOR THE PLUME AND SMOKE LAYER

The phenomenon of interest is the filling of an enclosure with smoke resulting from a combustion source near the floor as illustrated in figure 1. The buoyant plume, indicated by the wavy lines in the figure, entrains air as it moves toward the ceiling with a resulting decrease in the temperature and smoke concentration in the plume. The interface between the ambient air and the hot smoky region extending across the enclosure moves down at a velocity determined by the mass entrainment rate of the plume at the interface height. The interface will move down rapidly at first because of the high entrainment of the wide plume and then will move down ever more slowly toward the source. This qualitative description of enclosure "filling" is contained in the Baines-Turner[1] and the Zukoski[3] theories. The key results of these theories that pertain to the filling experiments will be presented below.
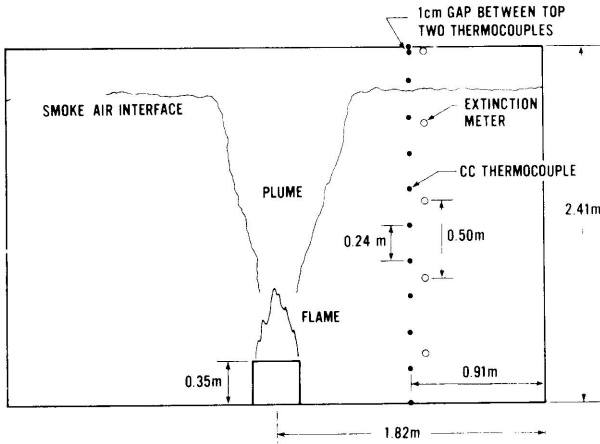

Fig.1 Schematic of the smoke filling experiment showing the positions of the burner, thermocouples, and extinction meters

The three quantities of major interest concerning the plume are the mass entrainment rate, $m_e(z)$, the temperature difference from ambient, $T(z) - T_o$, and the mass concentration of particulate, $\rho_p(z)$. The following results for the case of no thermal stratification are obtained from the Baines-Turner theory, which assumes that the density variation, $\delta\rho$, is a small fraction of the density, ρ.

$$\dot{m}_e = C_1 \alpha^{4/3} \rho_o \left(\frac{g}{\rho_o C_p T_o}\right)^{1/3} \dot{Q}^{1/3} z^{5/3}, \tag{1}$$

$$T - T_o = \frac{2\alpha^{-4/3}}{C_1} \left(\frac{T_o}{\rho_o^2 C_p^2 g}\right)^{1/3} \dot{Q}^{2/3} z^{-5/3}, \tag{2}$$

$$\rho_p = \frac{\alpha^{-4/3}}{C_1} \left(\frac{C_p \rho_o T_o}{g}\right)^{1/3} \dot{m}_p \dot{Q}^{-1/3} z^{-5/3}, \tag{3}$$

where $C_1 = \left(\frac{6}{5}\right)^{5/3} \left(\frac{5}{2}\pi^2\right)^{1/3}$.

The key parameters are the heat and particulate mass fluxes, Q and $\dot{m}_p$ respectively, and the height z measured from the burner surface. The quantity α is the entrainment coefficient, which has a value of about 0.1. The quantity g is the gravitational constant and the quantities ρ_o, C_p, and T_o refer to the ambient density, heat capacity, and absolute temperature of the air, respectively. The equations above are not expected to be valid near the source because of the high temperature and the resulting large value of $\delta\rho$ compared to ρ.

The Baines-Turner theory was originally expressed in terms of density difference, Δ, and buoyancy flux, F_o, which are related to the temperature difference and heat flux by the following relations:

$$\Delta = \frac{g(\rho - \rho_o)}{\rho_o} \quad -g\left(\frac{T - T_o}{T_o}\right) \text{ for } \frac{T - T_o}{T_o} \ll 1,$$

$$F_o = \frac{g\dot{Q}}{\rho C_p T_o}.$$

For gaseous plumes temperature and heat flux are the more convenient variables. The quantity $\rho_p(z)$, which is not treated in the Baines-Turner theory, is given in equation (3) assuming no particles in the surrounding air.

When the plume first reaches the ceiling, it spreads out across the ceiling and, provided the ceiling is not too high, the hot gas is assumed to move down vertically as a flat front. Baines and Turner observed this to be the case in their salt water experiments for H/R = 1 or less where H is the height of the enclosure and R the effective radius. For values of H/R = 1.5 and greater they found that the inertial force due to the arrival of the plume resulted in an overturning effect beginning from the outer part of the bottom layer.

Assuming no heat or particle loss to the ceiling or walls, one obtains from equations (2) and (3) with z set equal to H, the height of the enclosure, the temperature step and particulate concentration step at the layer interface. The validity of assuming an adiabatic ceiling will be considered below in light of our experiments.

Conservation of mass requires that the smoke front move downward as the ambient air is entrained into the plume. Thus,

$$\dot{m}_e + \rho_o A \frac{dz}{dt} = 0, \tag{4}$$

where A is the cross sectional area of the room. The mass flux from the source is neglected in equation (4). Substituting for $\dot{m}_e$ from equation (1) into equation (4) and integrating with respect to z, we obtain an expression for the time, τ, for the front to move from the ceiling to position z.

$$\tau = \frac{3}{2} \frac{\alpha^{-4/3} A H^{-2/3} \dot{Q}^{-1/3}}{C_1} \left(\frac{\rho C_p T_o}{g}\right)^{1/3}$$

$$[(H/z)^{2/3} - 1] \tag{5}$$

The predicted weak dependence of filling time on $\dot{Q}$ will be compared with experimental results. For a fixed room geometry and heat flux, the time for the layer to reach position z is predicted to vary as $z^{-2/3}$, which implies a slower and slower filling as the layer moves down, and this dependence on z will be compared with measured results.

2

Enclosures of interest in fire research typically have leaks and it will be assumed here that the pressure is constant within the enclosure. For the experimental situation in this study a leak is located near the floor. The filling time, τ, given by equation (5) does not include the effect of a leak near the floor. An analysis of the small effect of the leak on the filling time is given in Section 5.

In addition to the filling time, other quantities of interest during the filling process are the average temperature difference and smoke concentration in the upper layer and the temperature and smoke concentration profiles. Assuming a temperature change small relative to ambient, the average temperature difference can be related to the average density difference obtained by Zukoski[3] with the result

$$\frac{<T(z) - T_o(z)>}{\overline{T_o}(z)} = \frac{\dot{Q}t}{AH\rho C_p T(1 - z/H)} \tag{6}$$

where $< >$ indicates an average over z and $\overline{T_o}(z)$ represents the average initial temperature. The average smoke concentration, ρ_p, is simply the ratio of the total mass of particulate generated divided by the volume of the enclosure occupied by smoke.

$$\overline{\rho}_p = \frac{\dot{m}_p t}{AH(1 - z/H)} \tag{7}$$

As the smoke/air interface reaches the opening near the floor and the smoke starts to leak out, then equation (7) is no longer strictly valid. By calculating $\overline{\rho}_p$ as a function of time including the leak effect, it is found that $\overline{\rho}_p$ given in equation (7) is valid to within 10% for the following range of time, t:

$$t < \tau_F + \frac{0.2 \, AH}{\dot{V}} \tag{8}$$

where τ_F is the time for the smoke front to reach the leak, $\dot{V}$ is the flow rate out the leak, and the second term corresponds to the time required for the leakage of a volume of gas equal to 20% of the enclosures volume. For a 15 kW fire with 80% heat loss to the ceiling and walls in a 28 m^3 enclosure, this time corresponds to about 800 seconds, which is much longer than the duration of the experiments.

The final quantities of interest, the temperature and density profile, have not been calculated for arbitrary times. Baines and Turner[6] have determined the asymptotic long time limit for the density profile. For a small variation in density and an ideal gas, the fractional density difference is equal to minus the fractional temperature difference,

$$\frac{\rho - \rho_o}{\rho_o} = \frac{-\left(T - T_o\right)}{T_o} , \tag{9}$$

so the Baines and Turner result can be expressed in terms of an asymptotic temperature profile with the result:

$$\frac{T(z,t) - T_o}{T_o} = -\frac{\pi}{4}^{-2/3} (\rho^2 C_p^2 T_o^2 g)^{-1/3} \alpha^{-4/3} \dot{Q}^{2/3}$$

$$H^{-5/3} f_o(z/H) + \frac{\dot{Q}t}{AH\rho C_p T_o} \tag{10}$$

where $f_o(\zeta) = \zeta^{-2/3} (3.27 - 0.837\zeta - 0.062\zeta^2) - 2.37$

$$\tag{11}$$

The analogous equation for the mass concentration of particulate in the layer is given by:

$$\rho_p(z,t) = -\frac{\pi}{4}^{-2/3} \left(\frac{\rho C_p T_o}{g}\right)^{1/3} \alpha^{-4/3} \dot{Q}^{-1/3} \dot{m}_p H^{-5/3}$$

$$f_o(z/H) + \frac{\dot{m}_p t}{AH} . \tag{12}$$

These asymptotic solutions are obtained assuming no leakage or heat loss and a linear dependence of the temperature difference and particle density on time for long time. For the case of the equation of the particulate density, it is assumed that there are no particles in the ambient air and no condensation or wall loss.

INSTRUMENTATION AND PROCEDURE

The gaseous diffusion burner used in this study was constructed of a porous refractory material 0.30 m square. The temperature and velocity have been measured in the near field of this burner using natural gas as the fuel by McCaffrey[7]. As pointed out in the INTRODUCTION, an important consideration regarding the choice of this burner for this study is its repeatability in terms of heat flux and particulate mass flux. An indication of the overall repeatability of the burner output is the agreement within about $\pm$ 5% of the radiation output for duplicate runs.

The burner was positioned in the center of a room approximately 3.7 m square with a height of 2.4 meters. The top of the burner was 0.35 m above the floor so that the filling height was 2.05 m. The only opening in the enclosure was a 0.4 m high by 0.5 m wide orifice at the bottom and near the center of one wall.

In order to measure the movement of the smoke layer, twelve 0.3 mm diameter constantan-copper thermocouples and five extinction beams were approximately equally spaced from the ceiling to the floor at a position halfway between the burner and the wall as illustrated in figure 1. The uppermost thermocouple was pressed into the plaster board ceiling, the next 1 cm below and the remaining ten were spaced every 24 cm. The response time of the thermocouple, defined as the time for the temperature difference to drop by 1/e of the initial difference, was found to be 3.5 seconds based on a time-temperature measurement as the thermocouple--initially heated by 10°C--cooled in nominally quiescent air. This is fast enough for macroscopic quantities such as filling times, which are on the order of 30 seconds to 90 seconds, and average temperature profiles. The twelve thermocouples were calibrated at the steam and ice point, and when arranged in the test room, the thermocouple readings were found to agree within 1/2°C with a reference thermocouple. Small squares of aluminum foil 5 cm in length were placed as shields in front of the thermocouples to minimize the transport of radiant heat to the thermocouple.

The extinction meters used in this study were developed at Nohmi Bosai Company* and consisted of an incandescent light source, a collimating lens, a collecting lens, a pinhole, and a selenium photodetector. A typical Nohmi Bosai light beam was compared with a UL Standard extinction meter[8] for smoke generated by a heptane diffusion flame over a range of values for the extinction coefficient K from 0.05 to 0.69 m^{-1},

$$K = \frac{1}{L} \ln \frac{I_o}{I} , \qquad (13)$$

where L is the path length, I_o the incident light intensity, and I the intensity of the transmitted light. A plot of % transmission for UL extinction meter versus % transmission for the Nohmi Bosai meter yielded a slope of 0.969 compared to unity for the case of perfect agreement. A consistency check of the five extinction meters installed in the test room was made by mixing throughout the room highly concentrated smoke (K = 2.0 m^{-1}) generated by an acetylene flame. It was expected that design flaws of the light beam such as failure to eliminate scattered light entering the detector would be most evident at high smoke concentration. It was found that the deposition of smoke on the collecting lens of the extinction meter nearest the ceiling caused an additional attenuation of the light beam corresponding to as much as 10 to 12% of the total extinction coefficient; for the other four meters the deposition effect was 3-4% or less. After correcting for the smoke deposited on the lenses, it was found that all five meters agreed within 6%.

While in general the relation between an extinction measurement made with a polychromatic light source and the mass concentration of the smoke is complex depending on the size distribution and refractive index of the smoke, on the emission spectra of the light source, and on the spectral response of the detector, Lee and Mulholland[9] and Seader and Einhorn[10] have shown experimentally that the extinction coefficient is proportional to the mass concentration for smoke during flaming combustion. The study by Lee and Mulholland was concerned with soot generated by the combustion of hexane and measurements were made at low concentrations, 10 mg/m³, K = 0.08 m^{-1}, while the study by Seader and Einhorn included a variety of plastics--acrylo nitrile butadiene styrene (ABS), polyvinylchloride (PVC), urethane, and polystyrene--and measurements were made at concentrations as high as 1000 mg/m³, K = 8 m^{-1}. For all the materials studied, the following equation agreed with the data to within 10 to 20%.

$$K = \delta \, \rho_p , \qquad (14)$$

where ρ_p is the mass concentration of smoke in g/m³ and δ is a dimensional coefficient equal to 7.6 m²/g. We assume this relation to be valid for our work so

that the extinction coefficient is assumed to be a measure of the mass concentration of the smoke.

To start the experiment, the burner was ignited by a two meter pole one end of which was wrapped with cotton gauze, soaked in alcohol, and then ignited. Typically, the ignition pole was in position for only a few seconds except in the case of the lowest fuel flow rate; for acetylene fuel at a flow rate of 6.7 l/min the ignition time was 15 to 20 seconds. Upon ignition, temperature, smoke extinction coefficient and radiation output were recorded every five seconds.

RESULTS

Filling Time

Typical results for temperature versus time for the twelve thermocouples are shown in figure 2 for a 16.2 kW acetylene fire. The heat flux is calculated based on the flow rate of the fuel and its heat of combustion at 25°C to gaseous CO_2 and H_2O. The temperature of the gas 1 cm from the ceiling is monitored by thermocouple #2 while the temperature of the ceiling material, which is typically a few degrees lower than that of the gas as a result of heat conduction in the ceiling material, is monitored by thermocouple #1. The 3°C variation in the temperature with height at time t = 0 is a result of the ambient thermal gradient in the room. As an indication of repeatability, the value of the temperature difference T(z) - T_o(z) agreed to within about 10% at 60 seconds for the repeat runs.

The quantity of principal interest in this study is the passage time of the smoke front; that is, the time interval during which the interface between ambient air and the hot smoky region moves from the ceiling to the height of interest. According to the Baines-Turner theory, the smoke passage time should be discernible as the time at which there is a step change in the smoke concentration and temperature. As seen in figure 2, the temperature increases smoothly instead of the expected abrupt increase. In part this smooth increase is a result of the finite time constant of the thermocouple and of the radiant heating of the thermocouple, but, even in the case of the nearly instantaneous smoke con-centration measurements, which are represented in figure 3, the increase is not a step change though it is more rapid than in the thermal case.

An operationally convenient definition of the front passage time is the time at which the temperature difference at postion z has increased by a certain fraction, f, of a reference temperature difference. The reference temperature difference, ΔT_r, is based on the temperature of the thermocouple in the gas nearest the ceiling, thermocouple #2, at the transition from the initial rapid rise to the more gradual linear rise at later time. For example, for the experiment shown in figure 2 this transition point occurs at about 70 seconds with a temperature 22°C above the initial temperature. Times are taken relative to the time at which the temperature at position #2 first increases by $f \Delta T_r$.

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*Certain commercial equipment, instruments, or materials are identified in this paper in order to adequately specify the experimental procedure. In no case does such identification imply recommendation or endorsement by the National Bureau of Standards, nor does it imply that the material or equipment identified is necessarily the best available for the purpose.
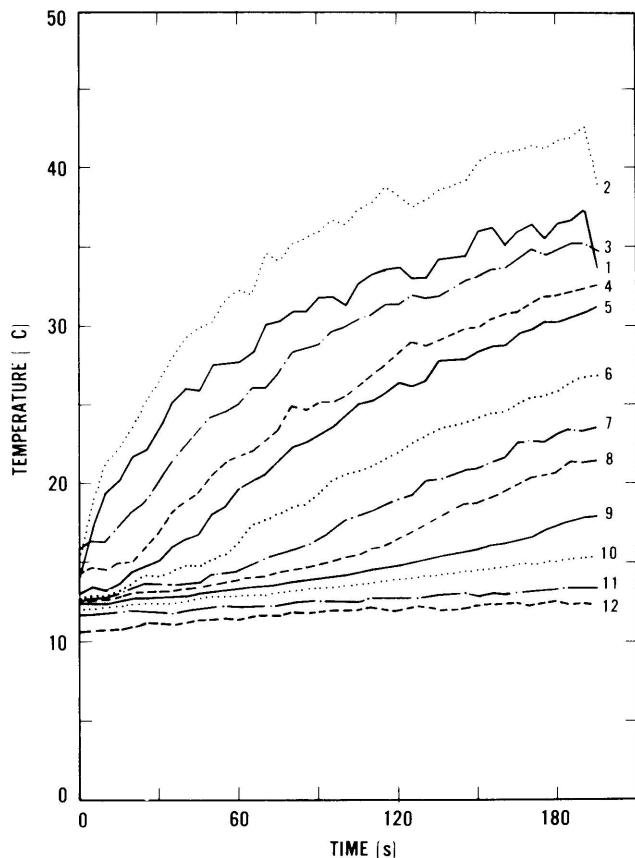
Fig.2 Temperature is plotted versus time for a
16.2 kW acetylene flame. Thermocouple 1 is
pressed into the plaster board ceiling, #2
is 1 cm below the ceiling with the remaining
thermocouples spaced approximately every
24 cm.

The passage times obtained by this method with
f = 0.1 are plotted in figure 4 for repeat experiments
with a 16.2 kW acetylene flame. The average variation
in the passage time for the six thermocouples was 10%
with the largest variations being for the shortest and
longest times. The five second interval between data
points accounts for some of the variation at the
shorter times while at long times the slowly increas-
ing temperature apparently becomes sensitive to
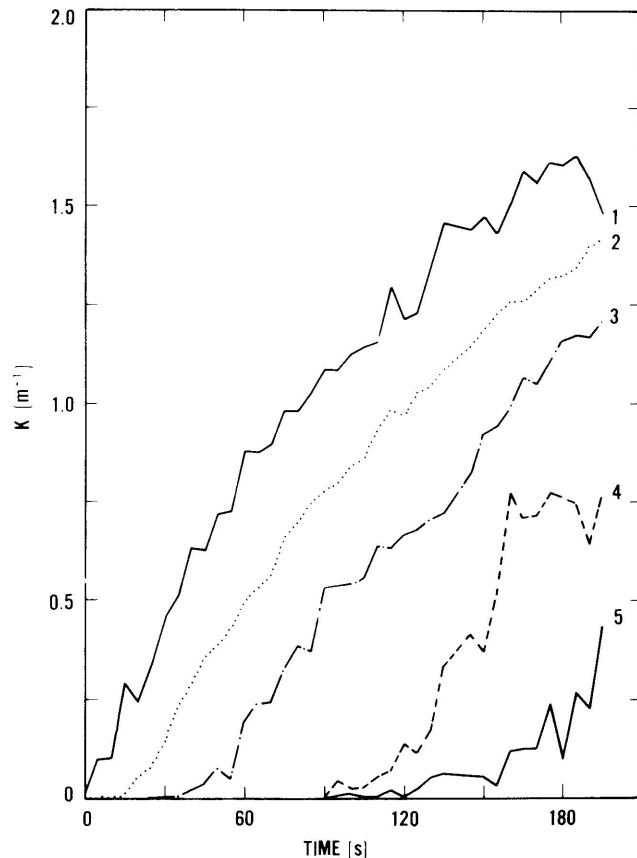effects not included in the Baines-Turner theory.

Fig.3 The extinction coefficient K of smoke is
plotted versus time as measured at various
heights (1, z = 2.09 m; 2, z = 1.55 m;
3, z = 1.05 m; 4, z = .55 m; 5, z = 0.05 m)
for a 16.2 kW acetylene flame.

The passage time versus height was also deter-
mined from measurements of the smoke concentration.
The variation of smoke concentration versus time is
is shown in figure 3 for a 16.2 kW acetylene flame.
The smoke concentration is monitored by five extinc-
tion meters spaced every 50 cm with #1 located at
the ceiling. The reproducibility is typically good
for the upper three extinction meters but not good
for the last two. In some experiments there was an
indication of smoke movement down the walls by the
early increase in smoke concentration at the lowest
meter position. The passage times based on smoke
concentration were determined by a method analogous
to the one used for thermal measurements, and as
demonstrated in figure 4 the passage times based on
smoke concentration and temperature typically agree
within 10 to 20% with the largest deviation for the
lowest position. Except for long times there is
good agreement between the data plotted in figure 4

and the theoretical prediction based on equation 5.
The value of α, 0.101, which is the one adjustable
parameter in equation 5, is based on the coefficients
(C_v and C_1) given by Zukoski[11] whose values are based
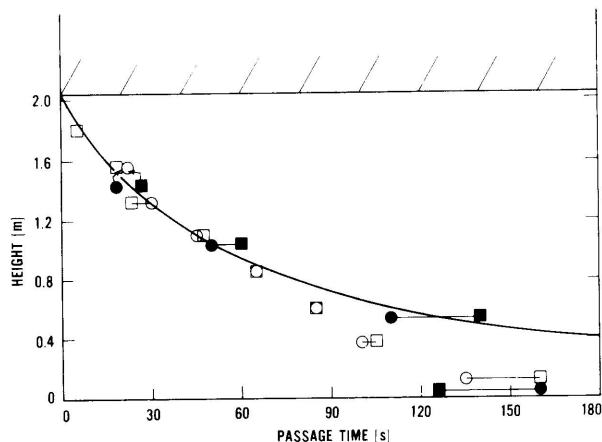on the experimental work of Yokoi[12].

Fig.4 The position of the smoke front is plotted
versus passage time for 16.2 kW acetylene
flames based on a fractional change f = 0.1.
Open symbols represent passage times based
on temperature measurements and solid symbols
are for passage times based on smoke extinc-
tion measurements. The different shape
symbols represent repeat experiments and the
curve corresponds to equation 5.

A more detailed study of the filling process was
made by observing the movement of a thin layer of
smoke. The layer was produced by first burning
propane fuel, which has a low sooting tendency, at a
heat output of 20.5 kW, then, without interrupting
the combustion process, quickly switching for about
30 seconds to acetylene fuel, which has a high sooting
rate, and then switching back to propane fuel. A
significant amount of soot is generated only during
the combustion of acetylene thus giving rise to a
thin soot layer analogous to the dye layer used by
Baines and Turner[1] in their salt water plume experi-
ments. The existence of a definitive smoke layer
moving from the top to the bottom is demonstrated in
figure 5 by the initial increase and then decrease
for the extinction meter near the ceiling, then for
the one 50 cm from the ceiling, and so on. The time
for the initial increase in smoke concentration
relative to the top extinction meter at the successive
positions agrees within 20% of the first front passage
times measured for 20.5 kW propane fuel as indicated
in Table I. This result is consistent with the find-
ings of Baines and Turner in their salt water experi-
ment that the agreement between the first front pas-
sage time and the passage time for a subsequent dye
layer was within 20% or better. Baines and Turner
also predicted that the two times be similar based
on a calculation of the layer passage time when the
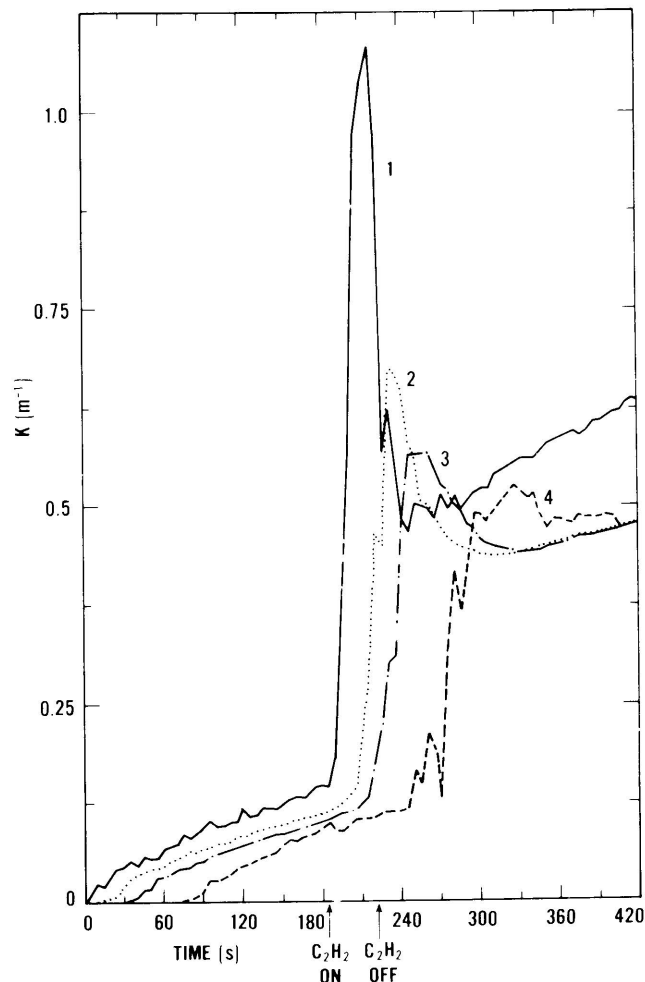plume and layer had reached their asymptotic state.

Fig.5 Extinction coefficient versus time shows smoke
layer moving downward from ceiling (1, z =
2.04 m; 2, z = 1.55 m; 3, z = 1.05 m;
4, z = 0.55 m). The smoke layer is produced
by switching from propane to high sooting
acetylene fuel for about 30 seconds.

TABLE I

Passage Time for Thin Smoke Layer and Smoke Front[a]

z,m	τ(layer),s	τ(front),s
1.57		20
1.55	23	
1.09		39
1.05	35	
0.61		79
0.55	78	

[a]Results based on the average of two repeat experi-
ments with propane fuel at 20.5 kW. In the thin
layer experiment the passage time is based on the
increase in concentration of acetylene generated
tracer while in the front experiment the time is
based on the temperature increase.

There are three features of the thin layer experiment not described by the Baines-Turner theory. The theory predicts a constant smoke concentration in the layer, but as indicated in figure 5 the peak concentration in the layer decreases by as much as 50% as it moves from the top to position 2 and by another 20% as it moves from 2 to 4. A second feature is the broadening of the peak as the layer moves down as indicated by the increase in transit time for the layer to go from above the light beam to below. This could be caused by an effective diffusion from the layer or an enhanced oscillatory motion of the layer as it moves down. Neither of these effects are included in the theory described in Section 2.

As is shown below Brownian diffusion of the smoke is too small to account for the observed broadening of the peak. An order of magnitude estimate of the broadening of the layer is given by the root mean square displacement of a smoke particle in the z direction, z_{rms}, which is related to the particle diffusion coefficient, D, by the equation

$$z_{rms} = \sqrt{2Dt}.$$

Using an upper estimate of the diffusion coefficient of the smoke particle, 10^{-4} cm^2/s, it is found that for time equal to the filling time of the enclosure, z_{rms} is less than 0.2 cm. The actual broadening of the layer is much greater, about 20 cm. The thermal diffusivity of air, 0.2 cm^2/s, is much greater than the particle diffusion coefficient and the characteristic thermal displacement for a time interval equal to the filling time is about 6 cm. Thermal diffusion may have a significant effect on the temperature profile at the smoke front; particulate diffusion clearly will not affect the smoke concentration profile.

The other feature, much less pronounced than the first two, is the apparent oscillatory behavior of the smoke concentration near the ceiling as if big globs of smoke were being periodically entrained into the plume. At least two additional peaks, much weaker than the first, could be observed for all three, layer experiments.

A limited number of experiments were performed with the burner located off center at a position halfway between the center of the room and the wall. (In this case, the distance from the burner to the thermocouples and extinction beams is doubled). As shown in figure 6 and 7, the off center case has a lower temperature and smoke concentration near the ceiling and a somewhat higher temperature and smoke concentration in the lower portion of the enclosure compared to the case with the burner in the center. The peak temperature in the off center case was recorded by thermocouple 3 located 24 cm (z = 1.81 m) rather than by the thermocouple nearest the ceiling (z = 2.04). The passage times, identified by the broad arrows in figures 6 and 7, are smaller for the off center case than for the center case. The times for half filling (z = 1.09 m) for the off center case are 0.35 to 0.65 of the center filling times as shown in Table II. There is an apparent discrepancy between these results and the Baines-Turner theory, which depends on the heat flux, the height, and cross section of the room but not on the position of the burner. One possibility is than the theory is incorrect in this regard. A second is that the theory does not apply because of an overturning effect resulting from the closeness

of the burner to the side wall. In the Baines-Turner analysis, the ratio of height to radius is used for assessing the significance of overturning, but perhaps the correct effective radius for the off center case is the distance to the closest wall and not the effective radius for the cross sectional area.
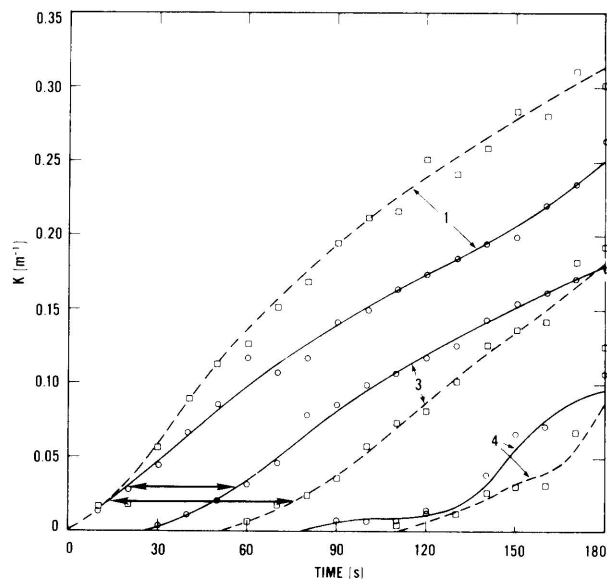

Fig.6   Smoke extinction coefficient is plotted versus time for the burner in the center of the room (□) and off center (○) for an 11.4 kW acetylene flame. The extinction measurements are made at three heights (1, z = 2.04 m; 3, z = 1.05 m; 4, z = 0.55) and the arrows represent the passage times.
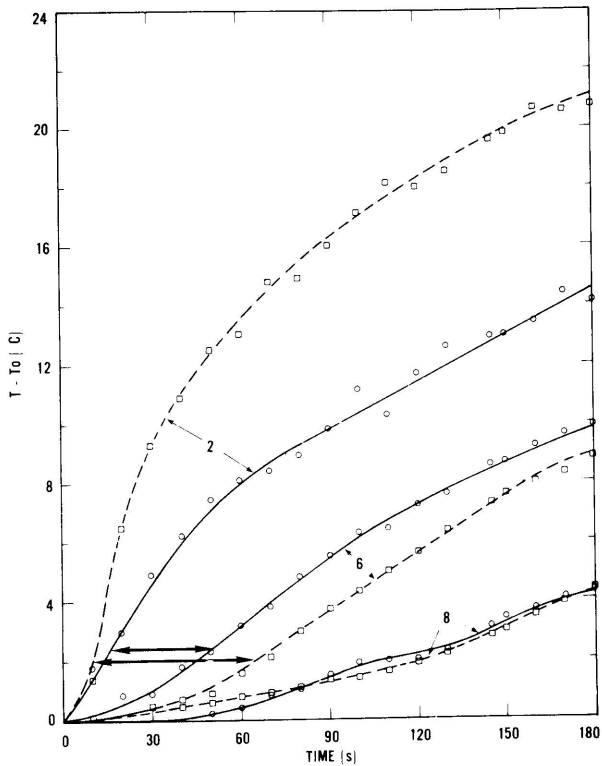
Fig.7 Temperature difference is plotted versus time for the burner in the center of the room (□) and off center (O) for an 11.4 kW acetylene flame. The temperature measurements are made at three heights (2, z = 2.04 m; 6, z = 1.09 m; 8, z = 0.61 m) and the arrows represent the passage times.

Temperature and Smoke Concentration Profiles

Typical results for a temperature and smoke concentration profile are shown in figures 8 and 9. A characteristic feature of the temperature profiles is a very steep temperature gradient near the ceiling in contrast to the more gradual gradient for smoke. This difference may in part result from significant heat loss to the ceiling and walls. After the steep drop near the ceiling, there appears to be a smooth temperature profile which at long times seems to be nearly linear over a majority of the room height. The smoke profiles, on the other hand, seem to be linear in the upper portion of room with small gradient and then drop rather abruptly in the lower portion of the room.
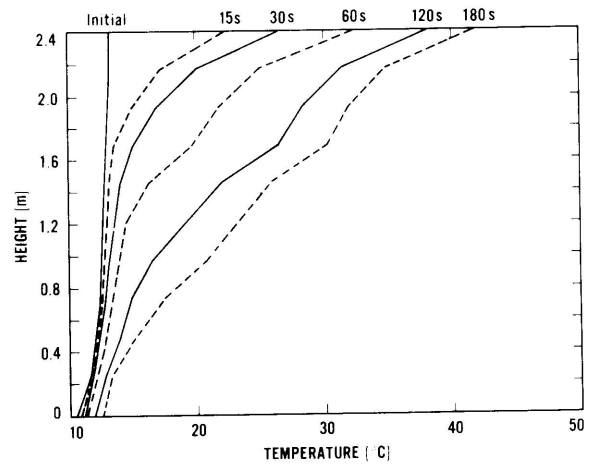

Fig.8 Temperature profiles are plotted for a 16.2 kW acetylene flame. In this plot z = 0 corresponds to the floor rather than the top of the burner, which is 35 cm above the floor.

TABLE II

Passage Time for Burner in Two Locations[a]

$\dot{Q}$	Fuel	τ(center), s	τ(off-center),s
11.4	acet.	49	32
16.2	acet.	46	
21.1	acet.	34	16
26.0	acet.	42	15
14.7	prop.	40	19
20.7	prop.	39	18
26.3	prop.	32	14
32.0	prop.	28	15

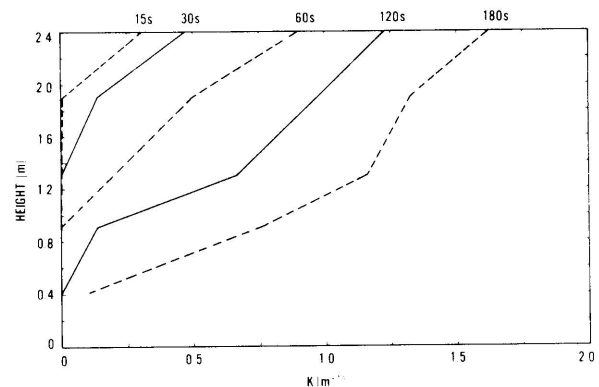[a]Time for layer to move from z = 2.04 m to z = 1.09 m based on thermal measurements.

Fig.9 Smoke concentration profiles are plotted for a 16.2 kW acetylene flame. In this plot z = 0 corresponds to the floor rather than the top of the burner, which is 35 cm above the floor.

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The temperature profile measured by Miyamoto et al.[4] 180 seconds after ignition for a 17.6 kW methanol fire is plotted in figure 10. The Miyamoto experiments were performed in an enclosure with a height of about 3.8 m above the burner and a square cross section with a width of 7.2 m. According to the geometric scaling factor in equation (5), a value of 180 seconds for the passage time to a fixed value of z/H for the large enclosure with the methanol flame corresponds to 70 seconds for the 16.2 kW acetylene flame in the smaller enclosure. The match in the temperature profile is good, but as shown in figure 11, for smoke concentration there appears to be a more uniform upper layer for the Miyamoto data. It is only the shapes of the curves that can be compared for the smoke data, since in one case the smoke is generated by the combustion of acetylene while in the other case "smoke flares" were positioned just above the smokeless methanol fire. It appears that the high value of smoke concentration near the ceiling in Miyamoto's experiment is an anomaly perhaps caused by smoke deposition on the extinction meter lenses.
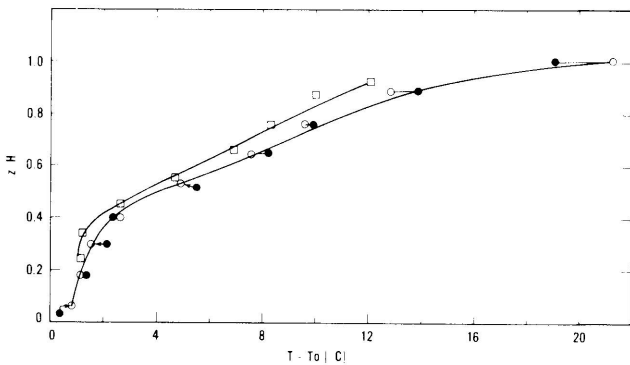
The only calculated profiles are the asymptotic long time results given by equations (10-13) and plotted in reduced form in figure 12. For comparing theory and experiment is is convenient to express equations (10) and (12) in the following form:

$$\frac{T(z,t) - T_o}{T_o} = -A_1 \, f_o(z/H) + B_1 t, \qquad (15)$$

$$\rho_p = -A_2 \, f_o(z/H) + B_2 t. \qquad (16)$$

Using the condition $f_o(1) = 0$, the quantity B_1 was obtained from the temperature-time slope; A_1 is obtained from the factors in equation 10. The data plotted in figure 12 correspond to profiles measured at 480 seconds, which is about 6 times longer than the 3/4 filling time. In the case of the smoke concentration data reduction, the smoke generation rate, which appears as a factor in the coefficient A_2, was determined from the measurement of ρ_p at z/H = 1 at 480 seconds. While it is difficult to draw any conclusion from the limited smoke data, it appears that the experimental value of f_o is much greater than the predicted value for small z/H. This could be a result of the leak in the enclosure allowing complete filling in the experiment, while this effect is not allowed for in the Baines-Turner theory. The agreement between theory and experiment is even worse for the temperature profile. An important effect not included in the theory is the heat loss to the ceiling and walls. Even if one treats $\dot{Q}$ as an adjustable parameter so that theory and experiment agree at one point, one still finds major differences in the shapes of the two profiles as shown in figure 12, where $\dot{Q}$ is chosen so theory and experiment agree at z/H = 0.14.

Fig.10 Temperature profiles are plotted for repeat 16.2 kW acetylene flame experiments (O , ●) as measured at 70 seconds and for Miyamoto's experiments with a 17.6 kW methanol flame (□) as measured at 180 seconds.

Fig. 12 The theoretical (solid curve) and experimental reduced profile f_o are plotted for 16.2 kW acetylene flame at 480 seconds. (● , smoke profile; ▲ , temperature profile; ■ , temperature profile with $\dot{Q}$ chosen so theory and experiment agree at z/H = 0.14) The dashed curves represent smooth fits to the data.
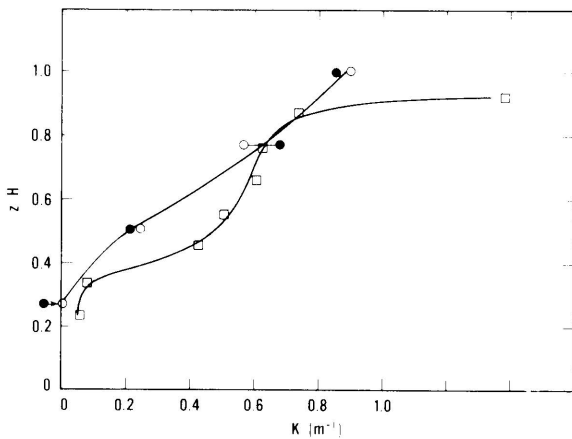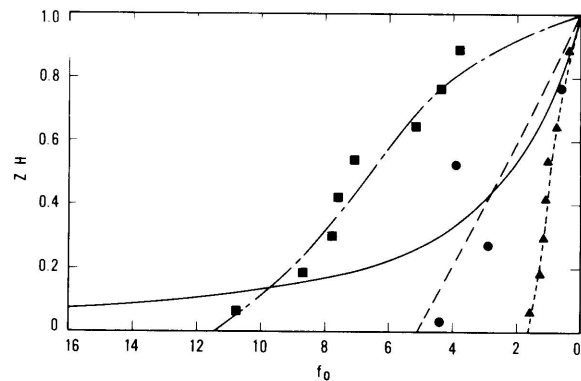

Fig.11 Smoke extinction coefficient profiles are plotted for repeat 16.2 kW acetylene flame experiments (O , ●) as measured at 70 seconds and for Miyamoto's experiments with a 17.6 kW methanol flame (□) as measured at 180 sceonds.

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It is useful to compare the predicted rate of increase of temperature and smoke concentration given by equations (6) and (7) with experimental results. The quantity $<T(z) - T_o(z)>$ was obtained for time t by locating the interface position and then obtaining the average temperature difference for all height elements in the upper layer. The first height element extends 24 cm from the ceiling to $z = 1.81$ m (from thermocouple 2 to 3), the next from $z = 1.81$ to $z = 1.57$ (from thermocouple 3 to 4), and so on all the way to the burner height. The temperature difference for each element was obtained as the average of the values for the two thermocouples bounding the element. There are eight elements above the burner height for the temperature measurement and four elements in the case of smoke concentration. The temperature and smoke concentration are assumed to be uniform throughout a cross section.

The experimental results for the time dependence of the temperature difference are plotted in figure 13 for propane flames with heat flux ranging from 14.7 to 32.0 kW, acetylene flames with heat flux ranging from 11.4 to 26 kW, and a 17.6 kW methanol flame. A large uncertainty is to be expected for small times because of the sensitivity of the term $(1 - z/H)$ to the thermocouple spacing. For example, a 12 cm error in the interface position, 1.57 m instead of 1.45 m, results in a 25% change in the reduced time. The linear relationship between the variables with all heat fluxes on one curve is consistent with equation 6, but the slope for the propane case and acetylene case are much less than the theoretically predicted slope. The slope for the methanol data of Miyamoto is smaller yet, but it is not to be compared with the theoretical curve in figure 16 because the enclosure size is much larger for the methanol fire.

In deriving equation (6) it is assumed that there is no heat loss to the ceilings and wall. If one assumes that the difference between theory and experiment is the result of heat losses, one can determine the effective value of heat flux, Q_{eff}, which would bring theory and experiment in agreement. From this analysis, it is found that over half the total heat is lost to the ceiling and walls: 68% for methanol, 78% for propane, and 84% for acetylene. Miyamoto et al.[2] estimated the total heat loss for their methanol pool fires using a different data analysis from the one suggested above; nevertheless, our value (68%), obtained from figure 11 of their paper for a time 180 seconds after ignition, agrees within 10% with their value (63%).

Because of the higher radiant heat flux of acetylene compared to propane, less of the heat of combustion is transmitted to the gas, and consequently the temperature rises more slowly than for propane as seen in figure 13. For 20 kW fire sizes with a 0.25 m diameter burner, McCaffrey[13] found that for propane 30% of the total heat was lost by radiation and 50% for the case of acetylene. Assuming these fractions to be valid for a 0.30 m square burner, the total heat flux transmitted to the gas can be obtained as the difference of the total heat flux minus the radiant heat flux. The ratio of the heat flux lost to the walls to the calculated heat flux transmitted to the gas is found to be 0.68 for both acetylene and propane flames.
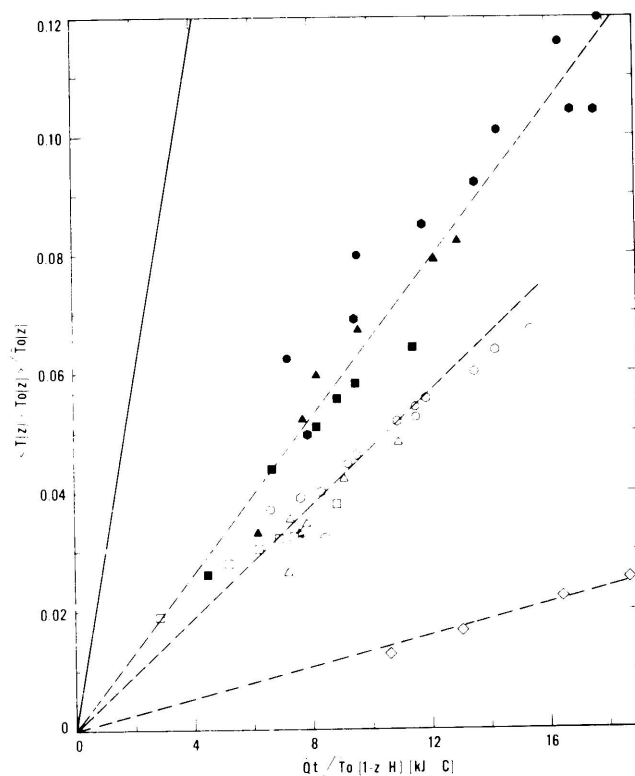

Fig.13 Average temperature difference is plotted versus reduced time for propane flames ($\bullet$ – 32.0 kW, $\blacklozenge$ – 26.3 kW, $\blacktriangle$ – 20.5 kW, $\blacksquare$ – 14.7 kW), acetylene flames (O – 26.0 kW, $\varnothing$ – 21.1 kW, $\triangle$ – 16.2 kW, $\square$ – 11.4 kW), and 17.6 kW methanol flame ($\diamondsuit$, Miyamoto). The predicted result based on equation (6) for an enclosure with height of 2.05 m and cross section of 13.6 m is shown by the solid line; the dashed lines represent linear fits of the data.

The nearly linear increase in the average temperature shown in figure 13 from 60 to 120 s suggests an approximately constant heat loss rate over this period of time. Another observation consistent with a constant heat loss rate is that the temperature difference between the top thermocouple in the ceiling and the thermocouple 1 cm below is nearly constant over this time period. The rate of heat transfer is proportional to the temperature difference so, at least at this one point, it is nearly constant.

The above analysis provides a rough estimate of the heat loss to the walls and ceiling. There is an uncertainty of about 10% based on the temperature measurements and fuel flow rate measurements. Implicit in this analysis is the assumption that the temperature is uniform throughout the cross section at a given height and that the plume region can be ignored. An extensive study with thermocouples positioned throughout the enclosure would be required to assess the uncertainty associated with these assumptions. For more precise analysis, the data for each fuel flow rate should be analyzed separately. McCaffrey[13] has shown the radiant loss fraction varies with the total heat flux; for example, for acetylene it varies from 0.49 to 0.46 as the heat flux is varied from 11.4 to 26.0 kW.

For the case of the average smoke concentration, $\bar{K}$, there is an approximately linear dependence on $t/(1 - z/H)$ as predicted by equation (7). The large scatter in the data shown in figure 14 is thought to be a result of the small number of extinction meters and the resulting coarse grain in the averaging, which is done by the same method as for the thermocouples. From the slopes in figure 14, the values of the mass generation rate of particulate can be obtained from equations (7) and (14). The strong dependence of the fraction of the fuel converted to particulate, $\dot{m}_p/\dot{m}_F$, where $\dot{m}_F$ is the mass flux of fuel, is illustrated in figure 15 for both propane and acetylene fuel. For acetylene about 10% of the mass of the fuel is converted to particulate at a flow rate of 25ℓ/min while for propane at the same flow rate the conversion is only 0.6%. There are a number of assumptions that go into the determination of the ratio in figure 15 including no wall loss of particles, uniform profile, and the simple relation between light extinction coefficient and smoke concentration given by equation (14). These could lead to systematic error as large as 50% for individual values though it is thought that trend is valid to a higher degree of confidence. The sigmodial shape of the curve with a region of very rapid increase in soot conversion with respect to fuel flow rate may be a general characteristic of soot generation in flames. Lee and Mulholland[14] observed this effect for laminar diffusion flames with ethylene, propane, and acetylene and Toossi[15] found a qualitatively similar effect for premixed acetylene/oxygen flame with the fuel/oxygen ratio for the premixed case playing the role that the fuel flow rate does for the diffusion flame.
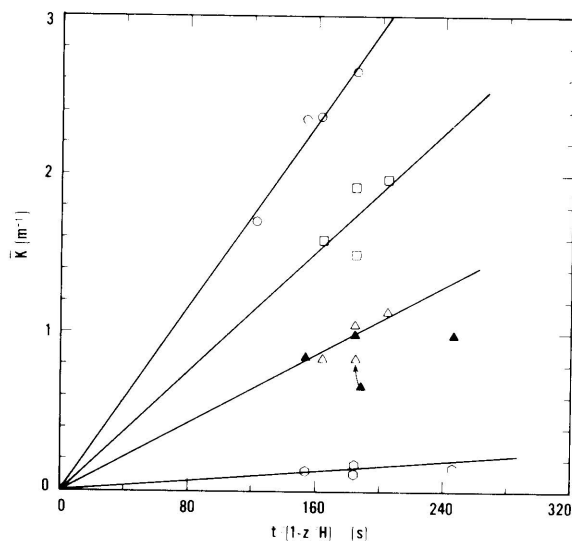

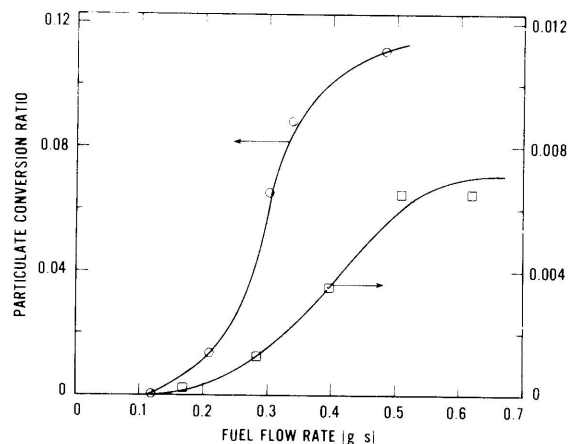Fig.15 Particulate conversion ratio, $\dot{m}_p/\dot{m}_F$, is plotted versus fuel flow rate for acetylene flames o and propane flames ☐ .

DISCUSSION OF SMOKE FRONT PASSAGE TIMES

The smoke front passage time data has been plotted in terms of reduced variables in figure 16 to facilitate the comparison between theory and experiments. According to Baines-Turner theory (equation 5), z/H versus $t\dot{Q}^{1/3}$ for experiments with a range of heat fluxes should reduce to a single curve for a given room geometry. The range in heat flux for the acetylene and propane data extends from 11.4 to 32 kW. The smoke front passage time was based on a fractional increase of 0.2 in the extinction coefficient for the acetylene smoke and a fractional increase of 0.2 in the temperature difference for the less sooty propane flame. The data were more consistent based on a fractional increase of 0.2 rather than 0.1. Most of the points on figure 16 correspond to the average of two repeat experiments. The variation in the passage time for repeat experiments was about 10 to 15% except for 3/4 filling and greater where the uncertainty is greater.
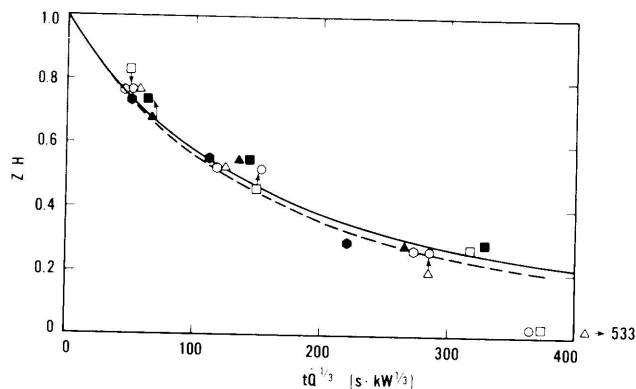

Fig.14 Average smoke extinction coefficient is plotted versus reduced time for acetylene flames (o - 26.0 kW, ☐ - 21.1 kW, ▲, △ - 16.2 kW, ◌ - 11.4 kW).

Fig.16 The smoke front position is plotted versus reduced time for propane flames (● - 26.3 kW, ▲ - 20.5 kW, ■ - 14.7 kW) and acetylene flames (◌ - 26.0 kW, △ - 21.1 kW, ☐ - 16.2 kW, o - 11.4 kW). The two theoretical curves correspond to no leak (——) and leakage with 25 kW flame (---). The front position is obtained by smoke extinction coefficient measurements for acetylene and by temperature measurements for propane.

It is seen from figure 16 that the reduced smoke front passage time is shortest at every height for the flame with the greatest heat flux compared to the one with the smallest for both acetylene and propane fuel. The range in the reduced passage times from the smallest to largest heat flux is about twice the uncertainty range for repeat experiments at the same heat flux. So the experimental results suggest that there is a small but detectable decrease in the reduced passage time with increased heat flux. Possible effects leading to this result include the leakage of fluid through the opening in the wall near the floor and an enhanced entrainment effect from a finite burner size.

An estimate of the effect of a leak on the passage time can be made based on Zukoski's[3] generalization of equation 4 to include the leak effect,

$$\dot{m}_e + \frac{(1 - \lambda)\ \dot{Q}}{C_p T_o} + \rho_o A\ \frac{dz}{dt} = 0, \tag{17}$$

where λ is the fraction of the heat lost to the ceiling and walls. The passage time τ is obtained from equation 17 by integration.

$$\tau = \frac{\dot{Q}}{\sqrt{g}\ H^{3/2} \dot{Q}_1 C_1 \alpha^{4/3}} \int_{\zeta}^{1} \frac{dx}{\left[x^{5/3} + \frac{(1 - \lambda)\ \dot{Q}_1^{2/3}}{C_1\ \alpha^{4/3}}\right]}, \tag{18}$$

where $\dot{Q}_1 = \dfrac{\dot{Q}}{\rho_o C_p T g^{1/2} H^{5/2}}$ and $\zeta = z/H$. The layer

height versus passage time obtained from equation 18 with $\lambda = 0.8$ and $\dot{Q} = 25$ kW is plotted in figure 16. It is seen that the leak effect decreases the 1/2 filling time by about 5% and the 3/4 filling time by about 10%. The observed decrease in filling time appears to be several times larger than this.

Concerning the effect of finite burner size, McCaffrey[7] found empirically the following relationship between the mass entrainment rate and heat flux in the near field of a plume:

$$\dot{m}_e \sim z\ \dot{Q}^{1/2} \tag{19}$$

This shows a stronger dependence on $\dot{Q}$ than predicted by plume theory ($\dot{Q}^{1/2}$ vs. $\dot{Q}^{1/3}$) and is in the correct direction to account for the enhanced $\dot{Q}$ dependence of the filling time. A quantitative analysis of the finite burner effect would require an expression for $\dot{m}_e$ over the full range in z.

The limited data available for comparison with our results include the studies of Miyamoto et al. and of Kawagoe. The enclosures used by Miyamoto, Kawagoe, and in our study had the following lengths, widths, and heights respectively: 7.2 x 7.2 x 3.8 m (Miyamoto), 7.3 x 4.7 x 3.0 m (Kawagoe), and 3.76 x 3.62 x 2.05 m (this study). In all three cases the ratio H/R is unity or less (0.91 to 0.99) assuring little effect from overturning. The geometric factor $A/H^{2/3}$ in equation (5) accounts for the longer passage times for the larger enclosures and the times in figure 17 are scaled by the geometric factor. For consistency the same definition of passage time was used in analyzing this data as that in figure 16 with Kawagoe's based on the measurement of the concentration of a smoke tracer and Miyamoto's on temperature measurements. There was an uncertainty on the order

of 10% in estimating passage times from the graph (Figure 11) in Miyamoto's paper[4]. In both studies there were no probes at the ceiling, and the position of the highest probe was taken as defining H. Since data was recorded every 60 seconds in Miyamoto's study, the smoke front position at 60 s was taken as H. In both studies methanol pool fires were used. The data of Miyamoto et al. is for a 0.25 x 0.25 m pool size while the Kawagoe data is for 0.30 x 0.30 and 0.50 x 0.50 m pool sizes. The $\dot{Q}$ used in figure 17 represents the total heat flux based on complete combustion.
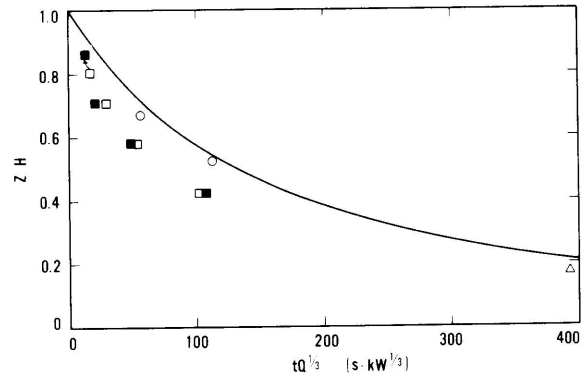
Fig.17 The smoke front position is plotted versus reduced time for Miyamoto's data with methanol flames (O - 17.4 kW) and for Kawagoe's data also for methanol flames (□ - 96 kW, ■ - 28 kW). The curve corresponds to equation 5.

The reduced passage times based on Kawagoe's study are only a fraction (0.5 to 0.6) of the times obtained in our experiments, while there is good agreement between the Miyamoto data and ours. No explanation for this large difference is known, though it is curious that the Kawagoe enclosure has a rectangular cross section (a length to width ratio of 1.6), while the other enclosures are nearly square in cross section.

CONCLUSIONS

Thermocouple and light extinction measurements of the passage time of the smoke-clean air interface for $\dot{Q}$ in the range 10 to 30 kW for a room size enclosure with an aspect ratio of 0.99 (H/R) are qualitatively consistent with theory though the dependence on $\dot{Q}$ is greater than predicted by the Baines-Turner theory. Experimental repeatability is on the order of 10 to 20% for the half filling time but markedly poorer as the front moves down. There is an indication of filling from below by smoke moving down the walls for the lowest position measured, 5 cm above the source.

It was found that the filling time depended on the burner location with a factor of two reduction in filling time observed for the burner located half way between the center and the wall. Either the Baines-Turner theory does not apply to the off-center case because of an overturning effect or else it is incorrect. In any event it is an area of interest to fire safety because of the central role of this theory in Cooper's[16] prediction of safe available egree time. There is a need for more study on passage time to

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quantify the effect of burner position and to provide a sound data base from which existing discrepancies in passage times studies can be understood.

Measurements of a thin layer of smoke appear to be a powerful experimental technique for studying the smoke filling process. Observations of the thin layer suggest either an appreciable diffusion effect from the thin layer or else an oscillatory behavior.

The measurement of the average layer temperature indicates that most of the heat of combustion is lost to the ceiling and walls, on the order of 80% in our experiments. The conversion ratio of fuel to particulate was found not to be constant with changing fuel flow rate, but rather to be non-linear with the ratio varying from less than 0.1% to over 10% as the fuel flow rate was increased from 0.12 to 0.50 g/s. The observed sigmoidal shape of the curve seems to be characteristic of particulate conversion ratio curves for combustion systems.

The long time smoke concentration profile does not increase rapidly with increasing height above the burner as predicted by the Baines-Turner theory for the asymptotic profile. This discrepancy may result from leakage or flow down the cool walls. The discrepancy between the long time temperature profile and the asymptotic profile is even greater, probably as a result of the heat loss to the walls and ceiling.

ACKNOWLEDGMENTS

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$$\Delta_o = 4^{-1} \pi^{-2/3} F_o^{2/3} \alpha^{-4/3} H^{-5/3} [f_o(\zeta) - \tau],$$

$$\Delta = 2^{-1} \pi^{-2/3} F_o^{2/3} \alpha^{-4/3} H^{-5/3} f(\zeta),$$

$$W = (\pi)^{-1/3} F_o^{1/3} \alpha^{-2/3} H^{-1/3} g(\zeta).$$

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